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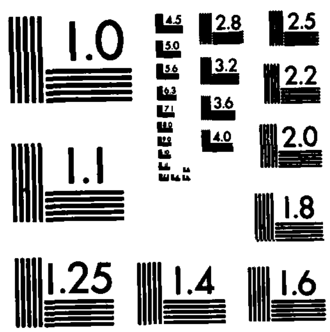

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ADVANCES AND TRENDS IN PLATE BUCKLING RESEARCH

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ADVANCES AND TRENDS IN PLATE BUCKLING RESEARCH*†

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SUMMARY

Recent advances and current trends in plate buckling research are summarized. The field is divided into three parts: (1) classical buckling studies, including plates of rectangular, circular and other shapes; (2) classical complicating effects, including elastic foundation, anisotropic material, variable thickness, shear deformation and nonhomogeneous material and; (3) nonclassical considerations, including postbuckling, imperfections, parametric excitation, follower forces, magnetoelastic buckling and inelastic buckling. Approximately 100 references for the period 1978-present are cited.

INTRODUCTION

The buckling of plates is a vast, complicated and somewhat disordered subject. Analytical solutions to problems began ninety years ago with the classical paper of Bryan (ref. 1) and have continued at a rapid rate since that time, yielding on the order of 2000 publications in the technical literature. Part of the achievement of earlier decades was summarized in the well known works by Timoshenko and Gere (ref. 2), Gerard and Becker (ref. 3) and Bulson (ref. 4). The writer is currently engaged in a research project attempting to organize, clarify and summarize this vast body of knowledge dealing with plate buckling by means of a comprehensive monograph. The intent of the present paper is to identify recent advances and current trends in plate buckling research from the relatively recent literature (i.e., 1978 - present).

CLASSICAL BUCKLING STUDIES

Classical buckling theory is based upon the assumptions of small deflections and a linearly elastic material, and yields the bifurcation behavior depicted in figure 1. That is, the well-known plot of transverse plate displacement versus inplane loading follows the ordinate (I) upwards, showing no displacement with increased load until a critical force (P_{cr}) is reached. At this bifurcation point the curve theoretically may continue up the ordinate (II) or, more realistically, may follow a buckling path (III), which is horizontal for the linear idealization.

The classical plate buckling problem is governed by the universally accepted differential equation of equilibrium:

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$$D\nabla^4 w + P_x \frac{\partial^2 w}{\partial x^2} + 2 P_{xy} \frac{\partial^2 w}{\partial x \partial y} + P_y \frac{\partial^2 w}{\partial y^2} = 0 \qquad (1)$$

in rectangular coordinates, where w is the transverse displacement, D is the flexural rigidity, ∇^4 is the biharmonic differential operator, P_x and P_y are compressive inplane forces (per unit length) and P_{xy} is inplane shearing force, along with appropriate linear equations representing the boundary conditions. This set of homogeneous equations defines a linear eigenvalue problem for which the eigenvalues are nondimensional buckling parameters, and the eigenfunctions describe the corresponding buckled mode shapes.

On the order of 1000 references can be found dealing with classical plate buckling. These treat a variety of shapes (e.g., rectangular, circular, triangular, parallelogram, sectorial), edge constraints (clamped, simply supported, free, elastically supported, intermittently supported, point supported), loading conditions (e.g., uniform and variable edge loads, body forces) and interior complications (e.g., holes, cracks, point supports). Exact solutions exist only for a very few problems involving rectangular and circular plates subjected to uniform normal edge loading. For all others solutions are necessarily obtained by approximate methods (e.g., Rayleigh-Ritz-Galerkin, finite elements, finite differences, collocation) the exact solution ultimately requiring the formulation of an infinite eigenvalue determinant. Approximate solutions are then obtained to any desired degree of accuracy by successive truncation of the determinant (and/or generating mesh size).

Rectangular Plates

Although hundreds of references are available for classical, bifurcation buckling problems for rectangular plates, the subject is far from exhausted (cf., refs. 5-17) and, indeed, quite a bit of useful information still needs to be found. The design considerations entering the problem include: (a) edge conditions (b) loading conditions (c) aspect ratio and (d) Poisson's ratio. For any arbitrary loading, there exist 108 combinations of classical boundary conditions (i.e., clamped, simply supported or free). Typically, one can find a significant amount of results in the literature for relatively few of these combinations (e.g., all sides simply supported, all sides clamped), and mostly for constant uniaxial or biaxial loading (i.e., P_x and P_y constant, and $P_{xy} = 0$ in eq. (1)).

The thorough development and widespread understanding of approximate analytical methods, along with the continued increase in digital computer speed and storage capabilities, have made it possible to obtain numerical results straightforwardly and cheaply which, not too long ago, were out of the question. Efficacy of this type was notably shown by Bassily and Dickinson (ref. 5) who used the Ritz method with beam functions to determine first the nonuniform inplane stresses present under certain loading conditions, and once again to calculate critical values of the loading parameters for buckling. The procedure was demonstrated with a cantilevered plate subjected to inplane acceleration loads (fig. 2). Kielb and Han (ref. 11) gave comprehensive results for hydrostatically loaded (i.e., $P_x = P_y$ = constant, $P_{xy} = 0$ in eq. (1)) plates having all six possible combinations of clamped and simply supported edges.

The most extensive results to date for rectangular plates were obtained recently by Kalro (ref. 10) using the Ritz method with algebraic polynomial trial functions. Complete sets of critical buckling loads were presented for rectangular plates having three aspect ratios ($a/b = 0.5, 1, 2$) and the following loading conditions:

(1) uniaxial compression (36 cases)
(2) hydrostatic compression (21 cases)
(3) constant shear stress (21 cases)
(4) inplane bonding (54 cases; e.g., fig. 3)

The results for shear loading may be seen in Table 1, arranged in order of descending buckling loads for square ($a/b=1$) plates.

Considerably more research has recently taken place on the buckling of rectangular plates having internal holes (refs. 18, 19) and cracks (refs. 20-22) than for the corresponding free vibration problems (ref. 23). Buckling mode shapes have also been determined experimentally by means of moire fringes (ref. 24).

Circular and Other Shapes

Not a great deal has been done recently to determine buckling loads and mode shapes for plates having non-rectangular shape. Circular plates have received some attention (refs. 25-28). Ku (ref. 27) developed a method for obtaining lower bounds for buckling loads, and used it on clamped and simply supported plates loaded in hydrostatic compression. Pardoen (ref. 28) demonstrated the finite element method on the same two problems, as well as on a clamped-free annular plate. Sato (ref. 29) analyzed elliptical plates utilizing the solution of equation (1) in elliptical coordinates in terms of Mathieu functions to solve problems for simply supported boundaries having elastic moment constraint.

Plates of parallelogram shape were analyzed by a number of researchers (refs. 30-36). Edwardes and Kabaila (ref. 30) discussed a method for dealing with the stress singularity that exists at the obtuse corners of simply supported plates by means of finite elements. The buckling of simply supported, isotropic plates was also studied by Kennedy and Prabhakara (refs. 31, 32) using a form of the series method; by Mizusawa, Kajita and Naruoka (refs. 33, 34) using the Ritz method with B-spline functions as trial functions; and by Thangam Babu and Reddy (ref. 36) using a finite strip method. Numerical results for the buckling parameters of regular polygonal, isotropic plates having 5, 6, 7 and 8 sides all clamped, subjected to hydrostatic compressive stress, were obtained by Laura, Luisoni and Sarmiento (ref. 37), using the Ritz method and algebraic polynomial trial functions.

CLASSICAL COMPLICATING EFFECTS

In this section a number of phenomena will be considered which serve to complicate the classical differential equation of the buckling problem (eq. (1)) and make it more difficult to solve. These complicating effects include:

(1) elastic foundation
(2) anisotropic material
(3) variable thickness
(4) shear deformation
(5) nonhomogeneous material, including laminated fibrous composites.

In each case, however, the resulting formulation typically still yields eigenvalue problems, and they are linear. The aforementioned phenomena arise quite commonly in practical design situations and their effects in buckling problems have all been studied for decades. Advances in understanding in each of these areas have been made during the past several years.

Elastic Foundation

In the case of a plate having its transverse displacement (w) resisted in both directions by a force supplied by an elastic foundation (or surrounding medium), the governing differential equation of equilibrium (1) is traditionally modified by adding a linear term, kw (where k is the stiffness of the foundation) to its left-hand side. The eigenvalue problem of free vibration is essentially unaffected by this added term, yielding frequencies whose squares are shifted upward linearly by the foundation stiffness (cf., ref. 38, p. 1). However, the plate buckling problem is typically a different one, requiring a more complicated solution of the governing differential equation.

Several recent works have appeared dealing with the effects of elastic foundations (refs. 39-43). Cimetiere (refs. 39, 40) examined the problem of the plate having unilateral constraints. Wang (ref. 43) used finite difference and finite elements to analyze the elastic buckling of an ice sheet, which is modeled as a semi-infinite plate on an elastic foundation with a semicircular cutout at the edge.

Anisotropic Material

For a plate composed of material which is generally anisotropic the first term ($D\nabla^4 w$) of the governing equation (1) is replaced by

$$D_1 \frac{\partial^4 w}{\partial x^4} + D_2 \frac{\partial^4 w}{\partial x^3 \partial y} + D_3 \frac{\partial^4 w}{\partial x^2 \partial y^2} + D_4 \frac{\partial^4 w}{\partial x \partial y^3} + D_5 \frac{\partial^4 w}{\partial y^4} \tag{2}$$

where $D_1, \ldots, D_5$ are constant coefficients depending upon the material properties. However, the second and fourth terms of expression (2) prohibit separation of variables and exact solution of the resulting differential equation. For plates of orthotropic material, with directions of rectangular orthotropy parallel to the edges, these terms vanish, and exact solutions are possible which are algebraic generalizations of the well-known ones for isotropic plates (i.e., plates having two opposite sides simply supported).

Considerable research has been reported in recent publications for the buckling of orthotropic plates (refs. 31, 32, 36, 37, 44-50). Of particular interest are the results for plates of non-rectangular shape, viz:

(1) parallelograms (refs. 31, 32, 36)
(2) regular polygons having 5, 6, 7 or 8 sides (ref. 37)
(3) polar-orthotropic annular plates (ref. 50)
(4) circular sectors having polar orthotropy (ref. 51)

Variable Thickness

Plates of variable thickness introduce considerable complication into the mathematical eigenvalue problem. In particular, the governing equation (1) must be generalized so that its first term ($D\nabla^4 w$) is replaced by a more general set of terms having variable coefficients, viz:

$$\nabla^2(D\nabla^2 w) - (1-\nu)\frac{\partial^2 D}{\partial y^2}\frac{\partial^2 w}{\partial x^2} - 2\frac{\partial^2 D}{\partial x\partial y}\frac{\partial^2 w}{\partial x\partial y} + \frac{\partial^2 D}{\partial x^2}\frac{\partial^2 w}{\partial y^2} \tag{3}$$

in rectangular coordinates (with similar complications in other coordinate systems, such as polar), where ∇^2 is the Laplacian operator and ν is Poisson's ratio. The flexural rigidity is defined, as in equation (1), by

$$D = \frac{Eh^3}{12(1-\nu^2)} \tag{4}$$

but now the thickness, h, is a variable. The resulting differential equation although linear has, in general, been incapable of exact solution.

However, if an energy approach is used, such as the Ritz method, no major complication is added by the presence of variable thickness. This method was recently used by Laura, Ficcadenti and Valerga de Greco to analyze the buckling of circular plates having a quadratic (ref. 52) and a piecewise linear (ref. 53) thickness variation. Gupta and Lal (ref. 54) were able to obtain solutions to the differential equation in polar coordinates by the method of Frobenius and used them to determine the critical buckling parameters for hydrostatically loaded, clamped and simply supported circular plates of linear thickness variation.

Shear Deformation

Consideration of shear deformation, in addition to the usual bending deformation, adds to the flexibility of a system, thereby decreasing the buckling loads and becoming of increased importance as the thickness-to-length (or width) of a plate increases. Mathematically, the theory is typically generalized to include three dependent variables, transverse total displacement (w) and orthogonal bending slope changes (ψ_x and ψ_y), yielding a governing set of differential equations of the sixth order (compared with the fourth order eq. (1)), and requiring three independent boundary conditions per edge. An application of a typical shear deformation theory to rectangular plates having two opposite sides simply supported was made by Hinton (ref. 55).

Ziegler (ref. 56) recently made a significant further development of thick plate theory including the effects of inplane (i.e., extensional) deformation together with shear deformation. He further showed that the inplane deformation effects can be more important than the shear deformation effects in the same problem.

Nonhomogeneous Material

Nonhomogeneous material may involve either continuous or discontinuous nonhomogeneity. The first case may readily arise, for example, for certain non-metallic materials such as rubber or styrofoam, or because of nonuniform thermal or other environmental effects upon the material properties, and includes material property variation through the thickness and/or in the other two directions. The second case includes layered, sandwich and fibrous composite plates. A number of references dealing with layered and sandwich plates have recently appeared (refs. 57-63).

Laminated fibrous composite plates have received a particularly large amount of interest in recent years. Here the geometry of the laminate (e.g., elastic modulus ratios of individual plies, angle-ply or cross-ply layups, number of plies, symmetry with respect to the midplane) is exceedingly important in determining stiffness properties of the plate and, hence, its buckling characteristics. For example, a cross-ply plate symmetrically laminated with respect to its midplane may be treated as an orthotropic plate. Similarly, a symmetric angle-ply plate may be accommodated by homogeneous, anisotropic plate theory. However, antisymmetric or asymmetric laminates result in bending-stretching coupling, resulting in an eighth order set of governing equations (as for a shell). For such plates it has been shown that buckling loads are significantly reduced from those predicted by homogeneous plate theory when only a small number of layers is used. A summary of research in the buckling of composite plates has recently appeared (ref. 64).

NONCLASSICAL CONSIDERATIONS

In this part considerations relevant to buckling will be treated, which are not dealt with by the foregoing linear, eigenvalue problems. These include

(1) postbuckling
(2) geometric imperfections
(3) parametric excitation
(4) follower forces
(5) magnetoelastic buckling
(6) inelastic material

Postbuckling

It is well known that, if a plate should reach its critical loading condition and buckle, under usual conditions it will deflect into a shape having finite amplitude which will be able to support the buckling load. This postbuckling configuration will correspond to another equilibrium state, and will typically require utilizing nonlinear, large amplitude plate equations in order to be determined analytically. Indeed, the plate will typically be able to withstand still larger loads by suffering larger displacements (curve IV in fig. 1) until it fails due to another reason, such as plastic collapse.

The subject of postbuckling has received significant consideration during the past four years (refs. 65-77). Numerical results for rectangular (refs. 66, 68, 69, 70, 72, 73, 75, 76), circular (refs. 66, 71, 77), annular (ref. 74) and parallelogram (ref. 67) plates have appeared.

Another phenomenon that may occur in the postbuckling range is called "secondary buckling". It has been shown both theoretically and experimentally that for loads sufficiently greater than the critical buckling value, and after significant postbuckling displacement has occurred, the plate may jump from one postbuckled configuration into another, having a different number of waves. Thus, curve IV of figure 1 may have a "secondary" bifurcation point. Indeed, still more bifurcation points may be reached as the load is increased further. This topic has received recent attention (refs. 68, 69, 72, 73, 75).

Geometric Imperfections

If a plate is not perfectly flat, then the application of small inplane forces at its edges or its interior will cause finite transverse diaplacements. This deviation from flatness (called an "imperfection") may be present in the unloaded state due to, for example, manufacturing or previous loading conditions, or it may be due to the presence of initial transverse loading or edge moments. The resulting inplane load-transverse displacement curve, is a nonlinear one. A representative plot is depicted by curve V in figure 1. That is, the curve deviates from the first part of the linear bifurcation path (I) as the load is increased, and then typically adapts itself to become asymptotic with the bifurcation postbuckling curve (IV) for large inplane loads. No bifurcation exists. As the imperfection amplitude approches zero, curve V smoothly approaches the kinked bifurcation path I-IV with sharply increasing curvature at the bifurcation point.

Recent studies of the effects of imperfections have included rectangular (refs. 78-80) and circular plates (refs. 81-84). Hui and Hansen (ref. 79) studied the infinite plate on an elastic foundation. Turvey (ref. 84) examined simply supported and clamped circular plates having linear thickness variation.

Parametric Excitation

In the case of "parametric excitation" part of the inplane forces are caused to vary periodically (usually sinusoidally) with time. Under some circumstances the forced vibration response will become dynamically unstable at smaller critical loads.

There has been a small amount of recent work on this subject (refs. 85-87). For example, Datta (ref. 85) made an experimental study on a thin rectangular plate having an internal slot with straight sides and circular ends, subjected to initial static tension in addition to the sinusoidal end force. As in the purely static case, local instability was observed in the vicinity of the slot. Tani and Nakamura (ref. 86) examined clamped annular plates having both edges subjected to the same static plus periodic radial loads.

Follower Forces

In classical plate buckling problems the inplane loads are prescribed to remain acting in the same plane while the plate deforms. If one prescribes that the direction of the force must follow the rotation of the plate at the point of force application, about an axis perpendicular to the force, then one has what is termed a "follower force". If the force only rotates some fraction of the total plate slope, then the fraction is called the "tangency coefficient". The resulting system is non-conservative with respect to energy and the analysis is a dynamic one.

Celep (ref. 88) recently studied the axisymmetric instability of a completely free circular plate having edge loads of arbitrary tangency angle. Farshad (ref. 89) examined the square cantilever plate and obtained results for the case of uniform compressive loads upon the two opposite free edges. Leipholz (ref. 90) treated the rectangular simply supported plate having tangential body forces of the follower type.

Magnetoelastic Buckling

A plate of magnetically soft material oriented so that its face is normal to a magnetic field may buckle as the field intensity is increased to a critical value. The destabilizing load may be a magnetic body torque which is proportional to the rotation of the plate at each point.

Miya, Hara and Someya (ref. 91) obtained experimental results for a cantilevered rectangular plate, and numerical results for solid and annular circular plates. Numerical results for the first problem were subsequently achieved (ref. 92). Van de Ven (ref. 93) examined circular plates having clamped, simply supported or free edges.

Inelastic Material

In this section we will consider plates made of material which, under the given loading conditions, violates the classical assumption of a linear elastic stress-strain curve. A typical and important case is that of elastic-plastic material behavior. In this case the inplane load-transverse displacement curves IV or V of figure 1 would begin to fall away from the curves shown when the load corresponding to the yield point is reached at any point in the plate. Another important case is that of creep buckling. Here the material undergoes time-dependent deformations. This case is not a stability problem of the classical type, but rather a matter of determining the length of time before failure.

Inelastic plate buckling problems have received considerable attention in the past four years (refs. 66, 94-114). A sample of the problems considered follows below. Dietrich et al. (ref. 96) made experimental, plastic buckling studies for simply supported rectangular plates subjected to biaxial compression. Gupta (ref. 100) developed a numerical procedure for the solution of plastic buckling problems, and demonstrated it for simply supported and clamped plates subjected to uniaxial compression. Popov and Hjelmstad (ref. 106) made tests of plate girder webs (which may be considered as plates) subjected to cyclic loading in the inelastic range. Needleman and Tvergaard (ref. 105) made analyses of the imperfection sensitivity of simply supported square plates exhibiting elastic-plastic behavior. Tvergaard (ref. 113) also considered creep buckling. Shrivastava (ref. 109) included transverse shear deformation effects in a plastic analysis of various uniaxially loaded rectangular plate configurations.

CONCLUSIONS

The field of plate buckling research is a reasonably active one, with more than 100 technical references having appeared in the past four years.

However, it seems that the rate of research using classical plate theory has ebbed somewhat in recent years, compared with the activity of previous decades. Certainly the 30 references found in the present search are far less than the 100 found for classical plate vibrations over a similar period (ref. 23). A similar disparity exists for classical complicating effects - less than 30 in the present work covering buckling, and about 200 for plate vibrations (ref. 116). Furthermore, the literature search for the present plate buckling paper was more complete than for the vibration ones (refs. 23, 116).

The number of classical buckling problems yet unsolved is still great. And because of their practical importance it is hoped that researchers will return to them in the coming decade and provide accurate and comprehensive results useful for design and to serve as a solid foundation upon which further nonclassical studies may be based. The analytical methods and computational capability used for plate vibration problems are equally useable for buckling solutions.

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TABLE I.-CRITICAL VALUES OF $P_{xy}b^2/D$ FOR INPLANE SHEAR LOADING

Edge Conditions	a/b		
	0.5	1	2
CCCC	404.89	144.50	101.20
CCCS	399.69	132.08	84.278
CSCS	397.48	124.06	66.232
CCSS	326.29	115.69	81.573
CSSS	319.83	105.69	65.285
SSSS	258.78	92.064	64.692
CCCF	323.66	83.997	27.260
CSCF	326.87	83.406	23.213
CFCF	319.10	74.175	17.509
CCSF	246.41	64.180	24.048
CSSF	247.54	63.665	19.146
CFSF	247.14	57.394	12.206
CSFS	90.308	49.027	45.605
SSSF	182.16	47.097	16.123
SFSF	174.60	41.823	8.0238
SFFF	38.452*	17.745*	5.0360*
FFFF	11.977*	9.9489*	2.9942*
CCFF	18.047	6.2201	4.5118
CSFF	17.896	4.8615	1.7462
CFFF	17.724	3.8833	.82613
SSFF	5.3476	2.6476	1.3369

*-First nontrivial eigenvalue (i.e., not a rigid body mode)

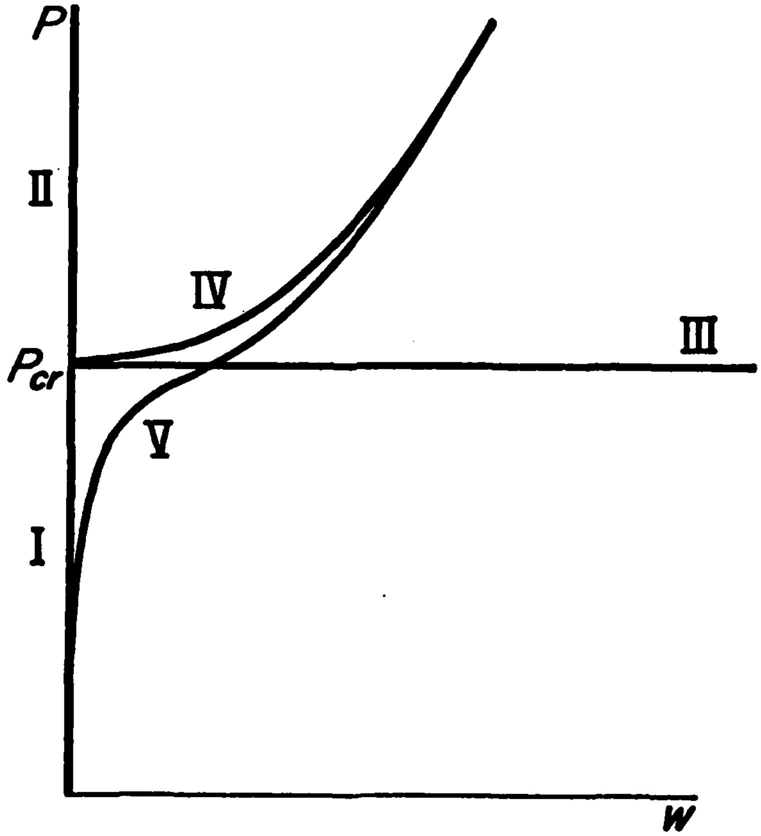

Figure 1.- Representative curves of load versus transverse displacement.

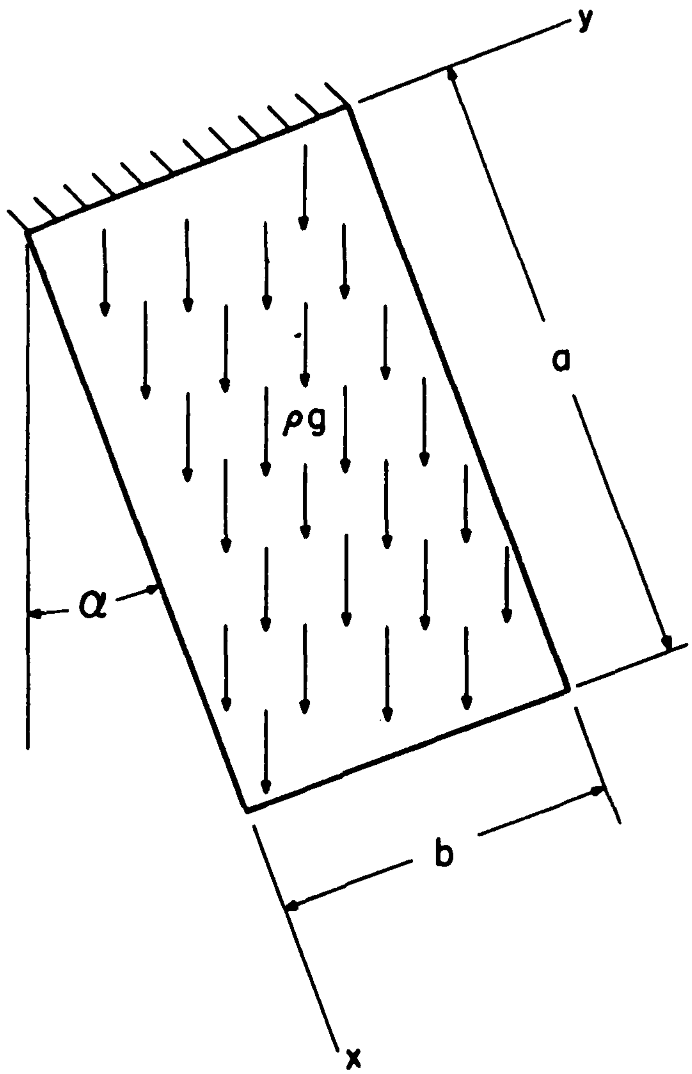

Figure 2.- Cantilever plate with inplane acceleration body forces.

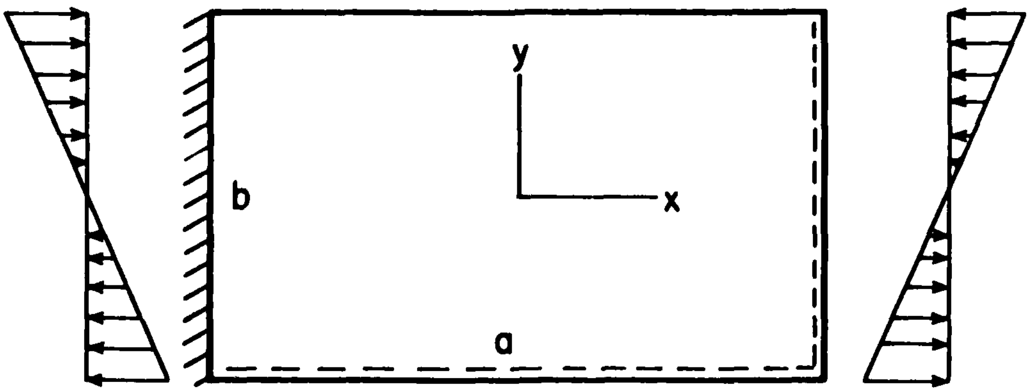

Figure 3.- Clamped, simply supported, simply supported, free (CSSF) rectangular plate with inplane bending stresses.

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